THE EFFECT OF EXCESS AIR ON FURNACE WALL TEMPERATURE AND HEAT TRANSFER FOR NATURAL GAS COMBUSTION IN A TUBULAR HEAT EXCHANGER

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CHAPTER I

Introduction

Some investigations have been conducted in the past years in analyzing the combustion and heat transfer in a central gas fired furnace. However, further research is still needed in order to clarify some of the factors which affect the heat transfer and fluid flow. The percentage of carbon dioxide content in the flue gas is one of such factors by which the mechanics of heat transfer from flame and combustion gas is influenced. Thus it was the objective of this study to determine how and to what extent this influence would be. In order to obtain a quantitative result, the following investigations have been made.

1. Measurement of furnace wall temperatures, its distribution, location of hot spots and their variations.
2. Measurement of flue gas temperature and its variation.
3. Determination of the amount of radiation from the combustion gas to the enclosing wall.
4. Determination of heat transfer coefficient from the combustion gas to the inside furnace wall.

In a heat furnace containing a flame, heat is generally transferred to the furnace wall in three ways.

(1) Heat is radiated directly from the combustion gas to the enclosing wall.

(2) Heat is radiated directly from the flame to the enclosing wall.

(3) Heat is transferred to the furnace wall by convection as the result of circulation of masses of heated gases in the furnace.

In order to obtain a complete analysis on this problem, an investigation was made by AGA and reported in its research bulletin 63(1). The model heat furnace used was of four inches diameter and four feet high. First the flow pattern in the vertical tube combustion chamber was separated into several regions by means of the detailed mathematical analyses as well as numerous observations and photographs of flames in the glass combustion chamber. From this developed a region or zone concept. With a particular flow observed in each region, an intimate analysis of heat transfer could be obtained. Generally, the heat furnace can be broken down into two main portions, the flame region, where wall temperature is increasing due to the heat release of the flame, and the cooling region where wall temperature is decreasing because combustion has ceased. Further separation of these regions gives a more detailed flow pattern as shown in Fig. 1.

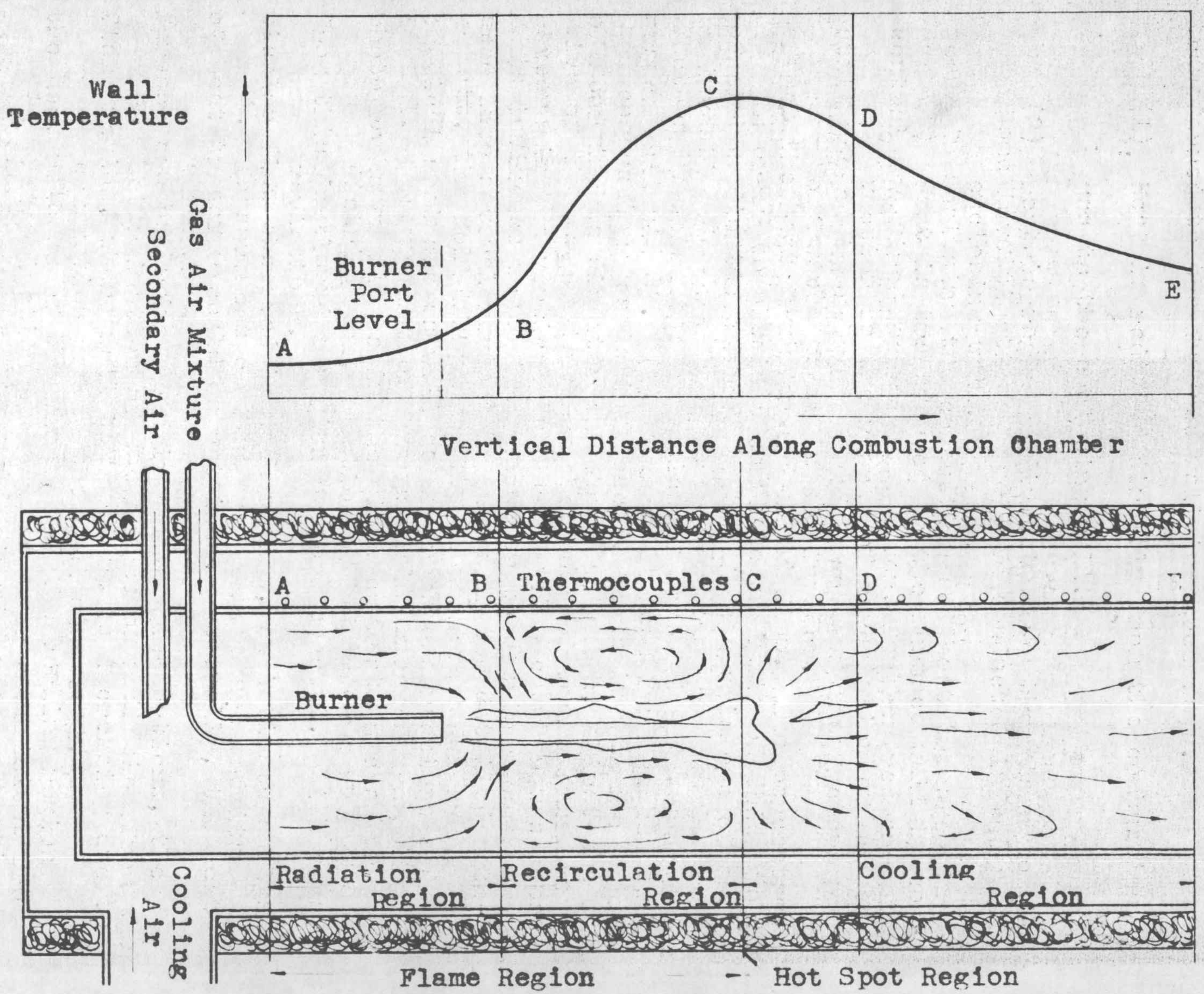


Fig. 1. Surface Temperature and Flow Patterns in Vertical Tube Combustion Chamber.

The radiation region occurs in the lower portion of the flame region in which a negligible amount of combustion product will be found. Heat transfer in this region is primarily by direct radiation. Since air is essentially transparent to radiation, it permits passage of radiant energy to the wall without absorbing any heat itself.

The upper portion of the flame region is named the recirculation region. It contains the flame jet and surrounding mixture of combustion gas and secondary air. The mixture forms a recirculatory motion which is drawn up along the boundary of the flame and then down along the furnace wall. Heat transfer in this region occurs by both radiation and convection.

The upper limit of the recirculation region is called the hot spot region. In this region the furnace wall temperature reaches a maximum. Actually the hot spot region is just a ring. The primary cause of the hot spot is the heated jet from the burner.

From the hot spot to the flue gas outlet is the cooling region. It contains a stream of combustion gases with occasional vortices and residual flames. About 80 per cent of the total heat transfer occurs in this region. Consequently, it is of great importance in the problem of heat transfer. Both radiation and convection take place in this region. But, because of the lower combustion gas temperature, convection plays a more important role than it does in the flame region.

In the present investigation, a vertical cylindrical furnace with three inch inside diameter and three foot high was employed. It was mounted inside a six inch diameter jacket to form an annular space in which the cooling air flowed. The heat input of 5000 BTU per hour was released by a Bunsen type atmospheric burner operating with natural gas. The excess air was varied from a range of 20 to 100 per cent in order to obtain different percentage of carbon dioxide in flue gas. The amount of primary air was also varied.

Because of the wide variety of furnace shapes and combustion situations in practice, the results concluded in this research probably are not applicable. However, it is the author's wish to delve deeper into the problem and provide more fundamental information that might lead to the improvement of gas furnace design and possibly to the building of more compact and simpler gas heating equipment.

CHAPTER II

Assumptions and Methods Used for Calculating the Heat Transfer and Its Coefficients

The region concept given by AGA described above gives a clear understanding about flow pattern in the vertical tube combustion chamber. The amount of heat transfer to the furnace wall was calculated for each region separately. But due to too many influencing factors, the problem was still complex. To simplify the problem, the region concept was ignored in the present investigation. The furnace was considered as a whole with heated uniform combustion gas flowing inside. Heat was considered to be liverated instantaneously at the entrance of the furnace. The cooling air flowing parallel outside the combustion tube made the whole system similar to a parallel flow heat exchanger. Heat transfer to the furnace wall could then be considered as simply due to the radiation from the combustion gas and convection resulting from the circulation of gas flow. The radiation from the flame was not considered because of its low percentage of total heat transfer. This assumption of neglecting the flame radiation was actually true. Because gaseous fuels provide nonluminous flames, the radiation directly from the flame was of small amount. In utilizing these assumptions for the calculation of heat transfer, it was necessary to treat the problem as if:

(1) The furnace contained only combustion gas and no flame was present.

(2) The fuel gas burned instantaneously at the inlet of furnace (burner port level).

(3) The combustion product and excess air were thoroughly mixed, the composition of the combustion gas was thus the same throughout the furnace.

(4) The heat transfer coefficient from combustion gas to wall and from wall to air was constant throughout the system.

(5) There were no insulation or other heat losses, all heat input was transferred to the cooling air except the flue loss.

In calculating the heat transfer by radiation, non-uniform distribution of combustion gas temperature was ignored. The whole system was assumed to be at a uniform mean temperature. Meanwhile, to evaluate the heat transfer coefficient from combustion gas to wall, the gas radiation was treated as a part of convection, so a combined heat transfer coefficient both for radiation and convection was determined.

As already pointed out, the source of radiation was only the combustion product or combustion gas itself at a uniform mean temperature, the heat transfer by radiation was thus simplified. Furthermore, because of the great difference in the emissivity or absorptivity of various gases, the diatomic gases such as O_2, N_2 and H_2 have very low emissivities, CO has a higher emissivity but it occupies a small percentage in combustion gas, hence only CO_2 and H_2O in the combustion gas need to be considered. Previous works show that the emissivity of gas mass in a furnace depends upon the temperature of the gas and the concentration of radiating molecules, i.e. the partial pressure of the constituent in the combustion gas and the radiation mean beam length due to the geometrical shape of the furnace. If more than one radiating constituent is present, the emissivities are additive, although a small correction should be made for interference of one type of molecule with the radiation from the other. Hottel and Egbert (4, p. 120-123) have given such experimental data and a detailed method to evaluate the emissivities can be found from other sources (3, p. 262-267 or 6, p. 82-86).

The net amount of radiant heat exchange between the combustion gas and the furnace wall is found by: (3, p. 266)

$$
\begin{aligned}
Q_r &= \sigma \epsilon_{w'} A \left[\epsilon_g T_g^4 - \alpha_g T_w^4 \right] \\
&= 0.1714\, \epsilon_{w'} A \left[\epsilon_g \left(\frac{T_g}{100}\right)^4 - \alpha_g \left(\frac{T_w}{100}\right)^4 \right] \qquad (1)
\end{aligned}
$$

where Q_r = The rate of heat transfer by radiation to the furnace wall from combustion gas, BTU/Hr

$\epsilon_{w'}$ = Effective emissivity of furnace wall, dimensionless

A = Effective heat transfer area of furnace wall, Ft^2

ϵ_g = Emissivity of combustion gas at T_g, dimensionless

α_g = Absorptivity of combustion gas at T_w, dimensionless

T_g = Mean temperature of combustion gas, ^oR

T_w = Mean temperature of furnace wall, ^oR

The slope of the wall temperature curve was used to calculate the heat transfer coefficient in the cooling region by AGA (1, p. 128). The same method was used in the present work to evaluate the combined combustion gas to wall heat transfer coefficient.

The difference between furnace wall temperature and ultimate system temperature (system temperature with no heat loss) was first plotted versus distance from the hot spot or any reference level on semi-logarithmic coordinate. This has been verified as a straight line if the assumptions in this investigation were true. The value of heat transfer coefficient was then solved from the expression

$$n = \frac{h_a h_g}{h_a \; h_g} \pi D \left(\frac{1}{w_a c_a} + \frac{1}{w_g c_g} \right) \qquad (2)$$

where n = Slope of the straight line, 1/ft

h_g = Combined gas side heat transfer coefficient, BTU/hr ft^2 ^{o}F

h_a = Air side heat transfer coefficient, BTU/hr ft^2 ^{o} F

D = Diameter of furnace, ft

w_a = Rate of cooling air flow, lb/hr

w_g = Rate of combustion gas flow, lb/hr

c_a = Specific heat of cooling air, BTU/lb ^{o}F

c_g = Specific heat of combustion gas, BTU/lb ^{o}F

The air side heat transfer coefficient was calculated by assuming all heat had been transferred to the cooling air except that carried away in flue gas. The relationship was found as (1, p. 124)

$$h_a = \frac{I - \text{flue loss}}{A\,(t_w - t_a)} \qquad (3)$$

where h_a = Air side heat transfer coefficient, BTU/hr ft^2 ^{o}F

I = Total heat input, BTU/hr

A = Area of furnace wall, ft^2

t_w = Mean furnace wall temperature, ^{o}F

t_a = Mean cooling air temperature, ^{o}F

By neglecting the conduction of the furnace wall, the overall heat transfer coefficient from combustion gas to cooling air could be calculated from the individual gas side and air side heat transfer coefficients. The quantitative relationship between overall heat transfer coefficient and its components is

$$\frac{1}{U} = \frac{1}{h_a} + \frac{1}{h_g} \qquad (4)$$

where U = Overall heat transfer coefficient from combustion gas to cooling air, BTU/hr ft^2 ^oF

h_a = Air side heat transfer coefficient, BTU/hr ft^2 ^oF

h_g = Gas side heat transfer coefficient, BTU/hr ft^2 ^oF

CHAPTER III

Experimental Apparatus

The apparatus used in this investigation was a simple tube in tube heat exchanger. Some associated equipment were provided to regulate the gas flow, meter the combustion and cooling air, measure the furnace wall and flue gas temperature and analyze the composition of flue gas. A general view of equipment is shown in the photograph of Figure 2.

The heat exchanger was a vertical cylindrical combustion furnace consisting of a three inch inside diameter pipe with height of three feet. The annular space for cooling air was provided by a six inch inside diameter steel pipe. To permit simple assembly of the exchanger, flanges were welded to the lower end of these pipes so that they could be bolted to the supporting frame in order. On the other end of the furnace, a three inch diameter cap was mounted. A one and one-half inch diameter pipe about one foot high was welded on the cap for the flue gas outlet. The inlet and outlet of cooling air were provided by welding two three inch diameter nipples horizontally on the lower and upper ends of the jacket with one on each side. The outside of the jacket was covered with one inch thick

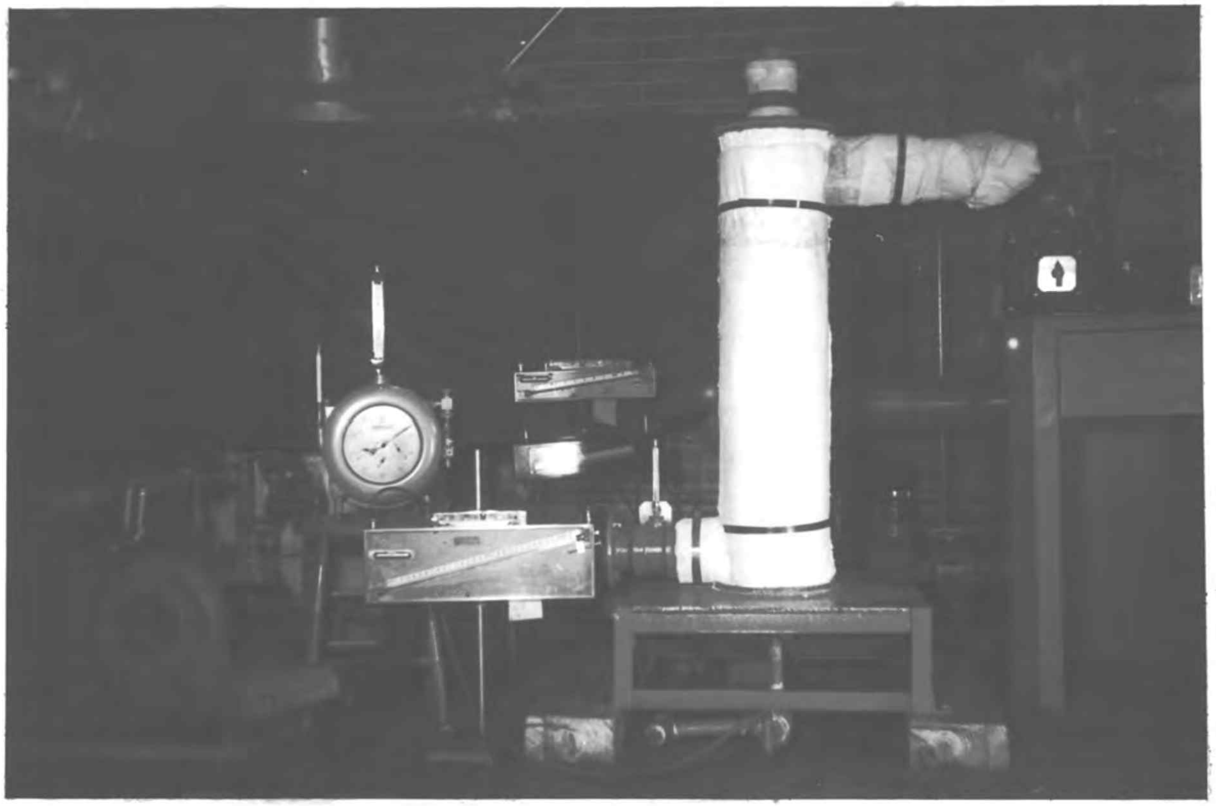


Fig. 2. General View of Heat Exchanger and its Associated Equipment

magnesia for insulation to reduce heat loss from the jacket wall. The arrangement of heat exchanger and the supporting frame are shown in Figure 3.

The burner was a single port three-quarter inch diameter Bunsen type atmospheric burner. A detailed drawing of the burner is indicated in Figure 4. The natural gas from the gas pipe passed through an orifice entered into the mixing tube in which the gaseous fuel was mixed with primary air before reaching the burner port. A straightening section was provided on burner head to retain the gas flame. In installing the burner into the furnace, care was taken to insure that the burner was placed centrally and axially. The burner port level was placed about five inches above the lower end of the combustion furnace.

The flow of gas was controlled by a globe valve upstream of the gas meter. A constant flow rate was verified by periodically checking the gas meter which was calibrated in the early stage of this investigation. Combustion and cooling air were supplied by two different size single stage centrifugal blowers. The rates of the flow were measured by the calibrated orifice meters installed on each line. The regulation of primary and secondary air was provided by their own globe valves. Figure 5 shows diagramatically the flow control method employed for both gas and air.

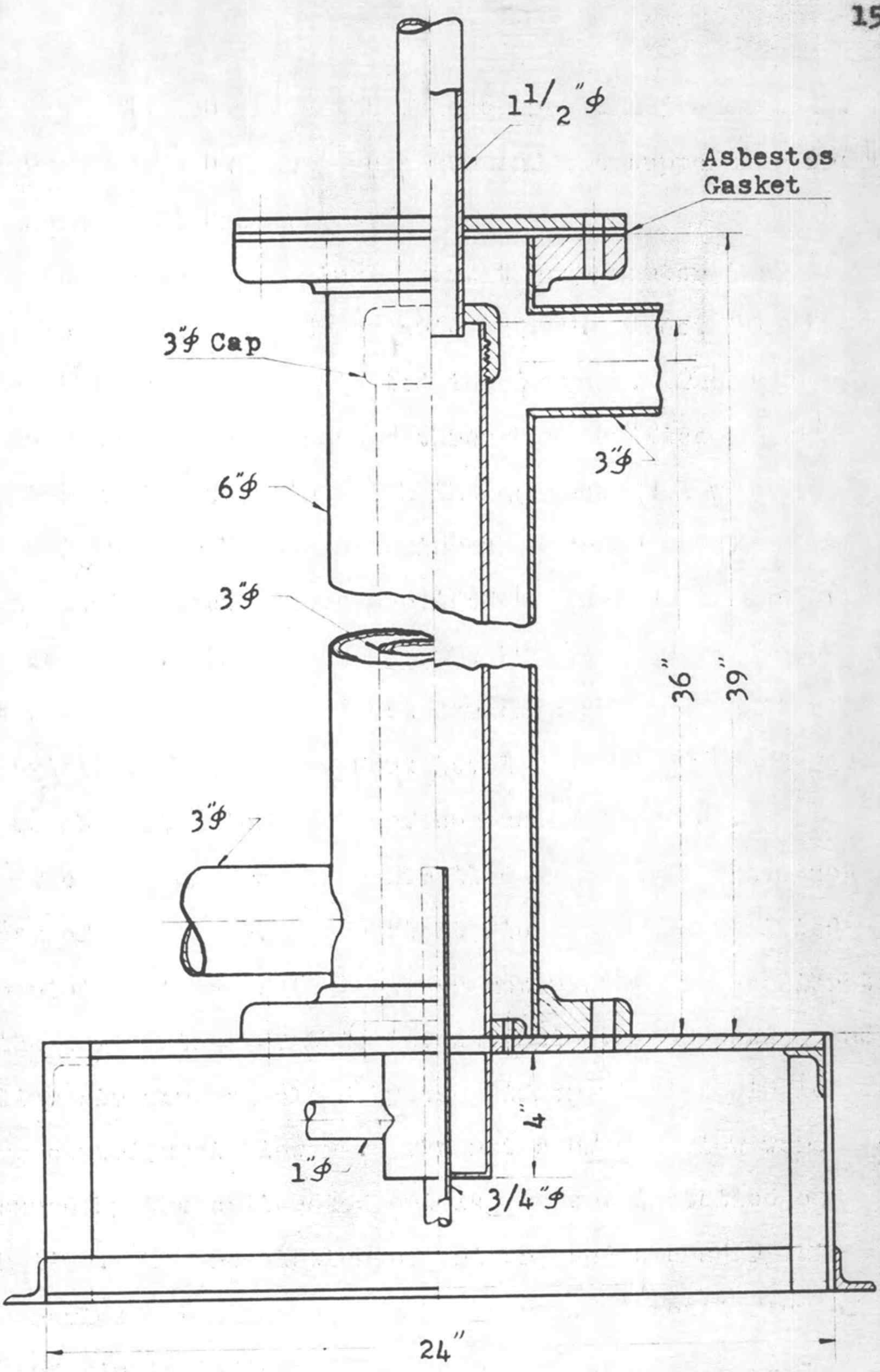


Fig. 3. Heat Exchanger and Supporting Frame

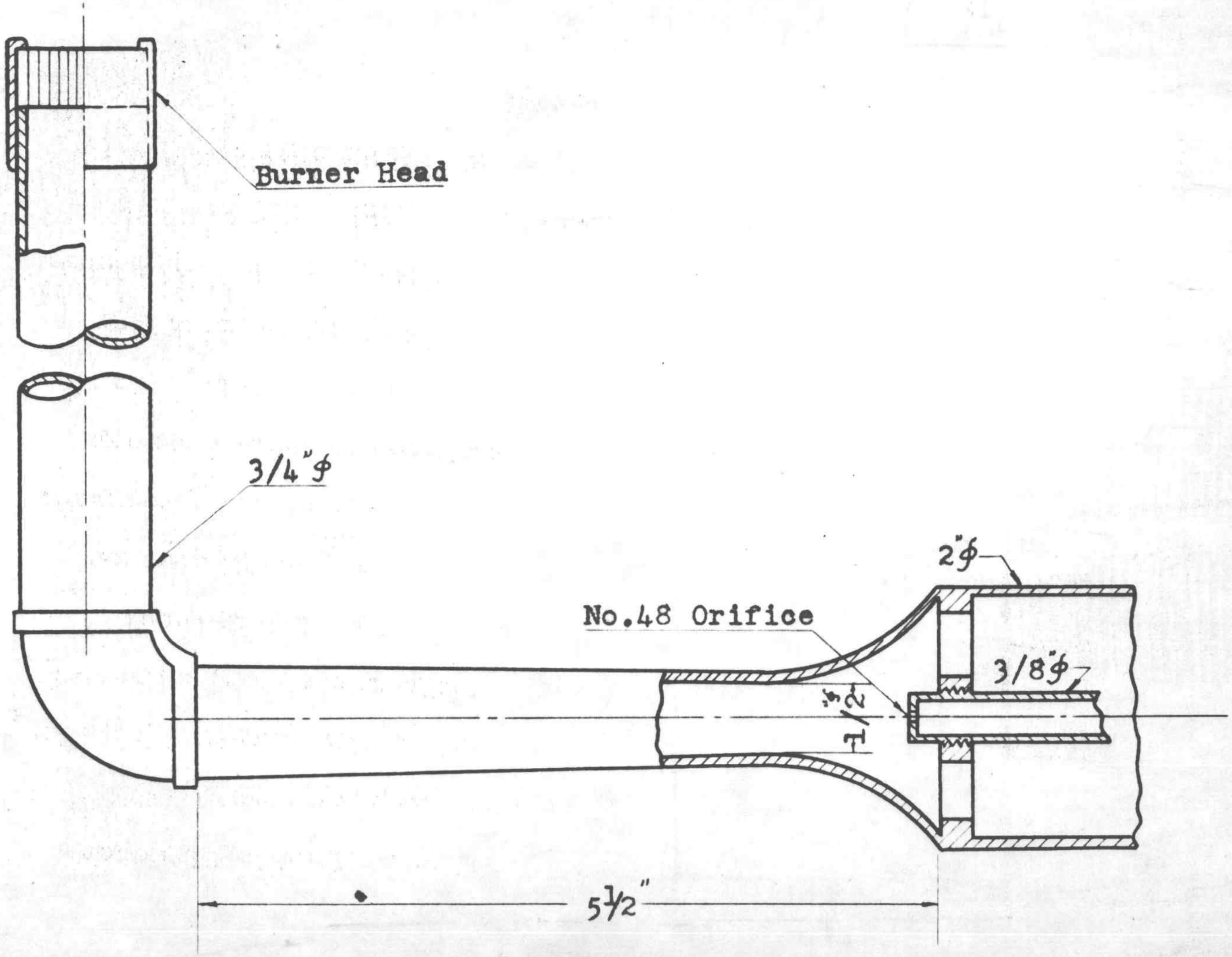


Fig. 4. Burner

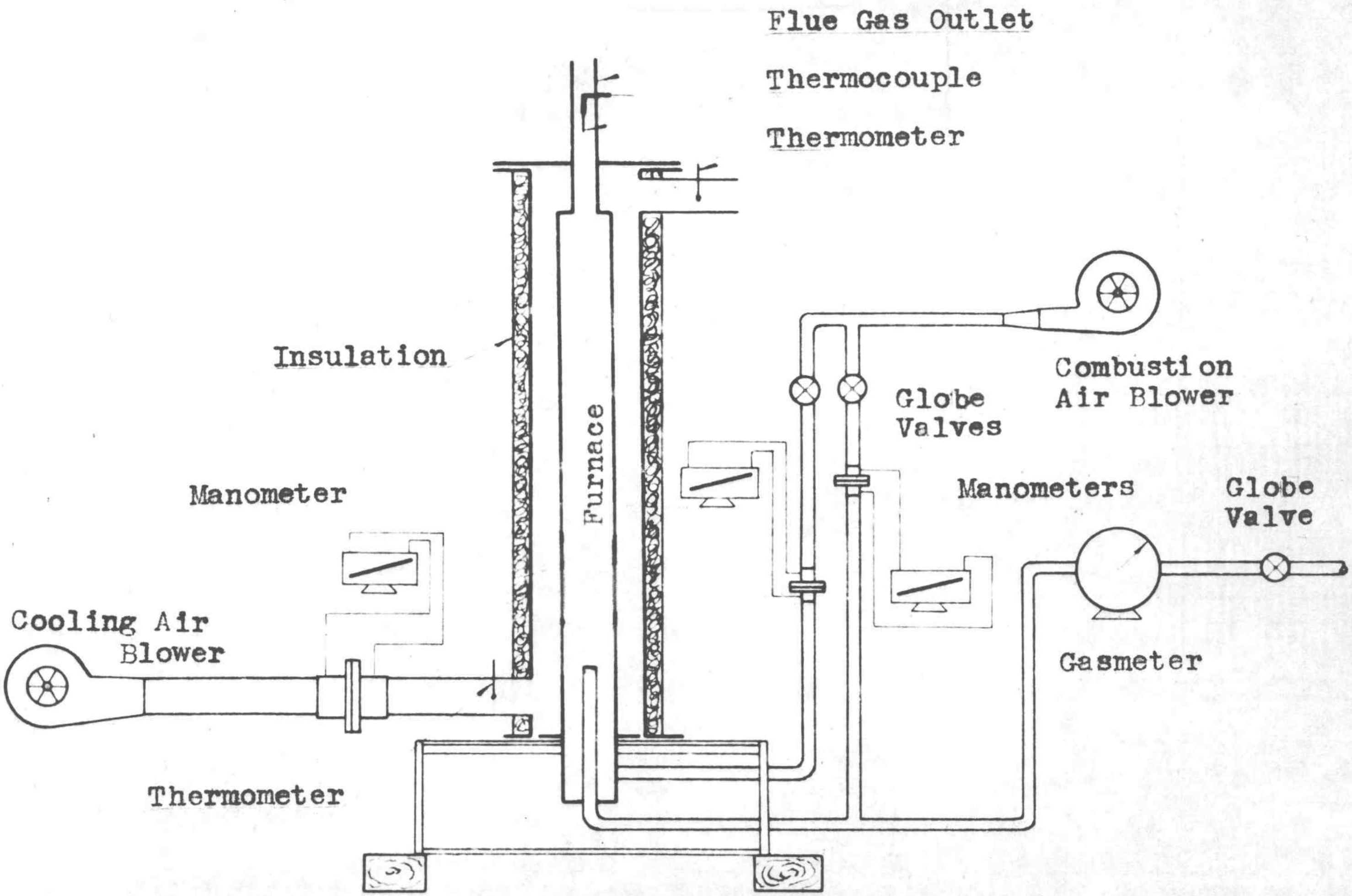


Fig. 5. Sketch of Heat Exchanger and Flow Controls

The measurement of gas wall furnace temperature proved to be most important from the standpoint of correctness. Therefore, in order to minimize the experimental error, the thermocouples were installed on the air side of the furnace wall to avoid direct exposure to the combustion gas and flame. The harness of leads from the thermocouple attachment were made small in bulk to reduce the disturbance of cooling air coming from below. Thermocouples used were No. 28 gage chromed alumel wire. Thirty-four thermocouples at one inch intervals were installed in position as shown in Figure 6. By passing along a section of cooling air outlet pipe, the lead wires were connected to two rotary selector switches which served as a cold junction. Temperatures were recorded as millivolts above cold junction temperature on a portable Brown potentiameter.

The No. 28 gage chromel alumel thermocouple wire was also used for measuring flue gas temperature. The portion close to the hot junction was bared and protected by two one-eighth inch diameter and two inch long double hole porcelain insulators. The two insulation tubes were placed at right angles to each other, one passing through a small hole in the outlet pipe wall while the other tube and the thermocouple wire were placed vertically downward. By moving the horizontal tube along the small hole, the

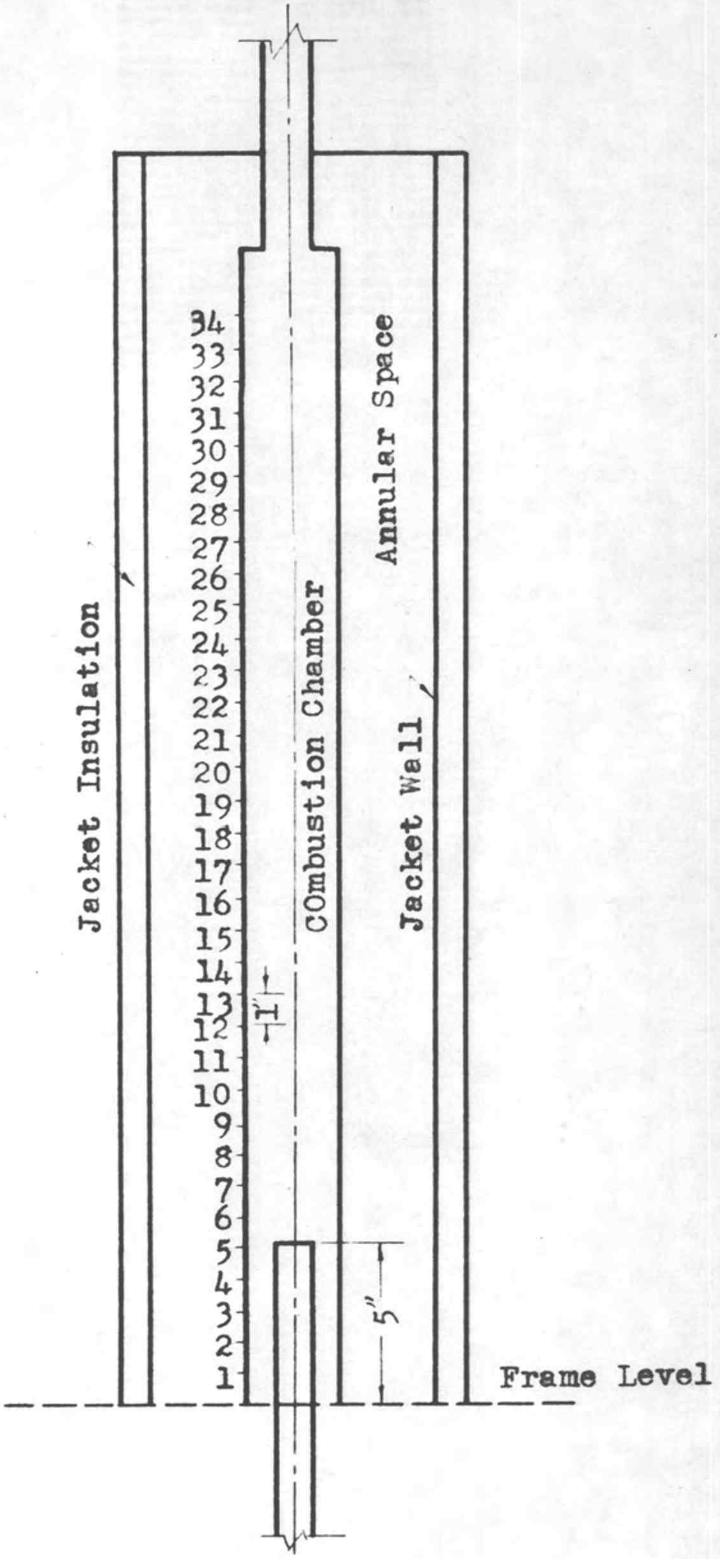


Fig. 6. Positions of Attachment of Thermocouple Wire

temperatures of flue gas at one-quarter inch diameter in the transverse place were measured. The tip of the thermocouple was shielded with an aluminum foil to reduce the error due to direct radiation.

The cooling air inlet and outlet temperature were measured by two mercury thermometers.

The flue gas samples were withdrawn by means of an aluminum tube and analyzed for CO_2 content and O_2 content by using an improved Hay's gas analyzer.

The relative humidity of cooling air and combustion air was taken by wet and dry bulb thermometers placed near the suction inlet of the air blowers.

CHAPTER IV

Experimental Procedures

The experimental procedures taken for this investigation were simple. The steps involved are as follows:

Heat input and combustion air were determined for the run.

Gas flow was adjusted until heat input was within one per cent of desired rate.

Primary and secondary air were adjusted.

Apparatus was operated at all desired flow rates with constant check until the hottest point on the furnace wall reached temperature equilibrium. Approximately one and one-half hours were needed from a cold start.

Condition of input and combustion air were recorded.

All thermocouple temperatures were measured with Brown potentiometer.

Dry and wet bulb temperature, cooling air inlet and outlet temperature and cooling air orifice meter reading were recorded.

Flue gas samples were withdrawn to the gas analyzer by means of a sample tube and an aspirator, percentage of CO_2 and O_2 content in flue gas were analyzed and recorded respectively.

If combustion air was changed in the run, about 20 minutes were needed to reach the new equilibrium.

CHAPTER V

Data and Calculations

An analysis of natural gas used in the present investigation gave:

Constituent	Per Cent by Volume
Methane	94.46
Ethane	4.25
Propane	0.93
Isobutane	0.07
n-Butane	0.08
Nitrogen	0.10
Carbon Dioxide	0.11
H. H. V.	1059 BTU/cu.ft.

Based on this, all calculated and experimental data can be presented below in tabulated form.

Table I. Amount of Theoretical Air Required for Complete Combustion

Constituent	Per Cent	Air Required# cu.ft./cu.ft. of Constituent	Air required cu.ft./cu.ft. of Gas
CH_4	94.46	9.528	8.990
C_2H_6	4.25	16.675	0.709
C_3H_8	0.93	23.821	0.221
C_4H_{10}	0.07	30.967	0.022
$n-C_4H_{10}$	0.08	30.967	0.025
N_2	0.10	-	-
CO_2	0.11	-	-
		Total	9.967 cu.ft./cu.ft. of Gas

#Data taken from 8, Chapter 6, Table 3, p. 118

Table II. Total Amount of Combustion Air

Excess Air Per Cent	Air Required cu.ft./cu.ft. of Gas	Air Required cu.ft./5000 BTU Input
20	11.96	56.4
40	13.95	65.8
60	15.95	75.1
80	17.94	84.6
100	19.93	94.0

Table III. H_2O Content in Combustion Gas

Aeration	Dry Bulb Temper-ature °F	Wet Bulb Temper-ature °F	H_2O Formed# by Combus-tion Cu.ft./hr	H_2O from Combus-tion Air Cu.ft./hr	Total H_2O Formed Cu.ft./hr	Amount of Combustion Gas Cu.ft./hr	Per Cent
40% Primary							
20% Excess	72	60	9.73	0.76	10.49	61.12	17.3
40% "	73	60	9.73	0.86	10.59	70.52	15.0
60% "	73	60	9.73	0.98	10.71	78.82	13.4
80% "	73	60	9.73	1.10	10.83	89.32	12.1
100% "	74	60	9.73	1.20	10.93	98.72	11.1
60% Primary							
20% Excess	75	59	9.73	0.63	10.36	61.12	16.9
40% "	75	59	9.73	0.73	10.46	70.52	14.8
60% "	74	59	9.73	0.86	10.59	79.82	13.3
80% "	74	59	9.73	0.97	10.70	89.32	12.0
100% "	74	60	9.73	1.08	10.81	98.72	11.0

#Calculated from H_2O =

$$(2.0 \times CH_4 + 3.0 \times C_2H_6 + 4.0 \times C_3H_8 + 5.0 \times C_4H_{10} + 5.0 \times n\text{-}C_4H_{10}) \times \frac{5000}{1059}$$

Cu.ft./hr

Table IV. CO_2 Content in Combustion Gas

Aeration	% CO_2 Content Analyzed from Flue Gas	% CO_2 Content on Wet Basis
40% Primary		
20% Excess	9.2	7.84
40% "	7.8	6.78
60% "	6.8	6.00
80% "	6.0	5.35
100% "	5.3	4.77
60% Primary		
20% Excess	9.5	8.12
40% "	7.8	6.80
60% "	6.6	5.83
80% "	6.1	5.45
100% "	5.5	4.95

Table V. Percentage of Flue Loss

Aeration	% CO_2 Content in Flue Gas	Average Flue Gas Temp. °F	Room Temp. °F	t °F	Flue# Loss %
40% Primary					
20% Excess	9.2	475	72	403	19.6
40% "	7.8	500	73	427	22.0
60% "	6.8	523	73	450	24.5
80% "	6.0	545	73	472	27.2
100% "	5.3	563	74	489	29.5
60% Primary					
20% Excess	9.5	461	75	386	19.0
40% "	7.8	492	75	417	21.7
60% "	6.6	514	74	440	24.5
80% "	6.1	534	74	460	26.5
100% "	5.5	551	74	477	28.6

#Taken from 8, Chapter II, Chart 9, p. 281.

Table VI. Furnace Wall Temperature Data

Aeration	Temperature °F Vertical Distance Along Furnace Wall, Inches											
	1	2	3	4	5	6	7	8	9	10	11	12
40% Primary												
20% Excess	102	113	123	125	163	167	204	279	344	365	382	411
40% "	102	112	124	124	170	184	198	269	332	360	372	407
60% "	100	110	118	120	152	176	186	248	303	332	340	382
80% "	98	108	116	117	148	170	179	235	286	313	320	360
100% "	99	108	115	116	145	165	172	224	271	299	304	345
60% Primary												
20% Excess	107	118	131	134	174	199	232	295	358	373	394	418
40% "	106	117	128	130	171	196	219	289	348	371	390	419
60% "	100	114	125	126	162	186	206	274	335	356	370	408
80% "	100	111	119	121	155	181	195	252	308	332	344	386
100% "	99	109	117	118	150	173	185	237	287	311	322	362

Table VI. Furnace Wall Temperature Data (continued)

Aeration	Temperature °F Vertical Distance Along Furnace Wall, Inches											
	13	14	15	16	17	18	19	20	21	22	23	24
40% Primary												
20% Excess	449	447	442	427	406	403	388	370	368	356	345	335
40% "	445	444	440	423	400	396	382	365	363	351	342	333
60% "	432	429	428	415	395	394	378	359	357	349	340	331
80% "	402	410	414	405	387	386	371	355	355	342	335	326
100% "	388	395	402	396	382	381	370	353	352	342	335	325
60% Primary												
20% Excess	452	452	451	438	420	418	403	384	382	369	354	343
40% "	449	449	447	431	409	406	392	374	374	359	348	340
60% "	439	437	431	415	391	387	372	355	355	345	336	328
80% "	423	426	425	409	390	385	370	354	354	342	333	326
100% "	400	407	410	399	381	380	366	349	349	338	329	322

Table VI. Furnace Wall Temperature Data (continued)

Aeration	Temperature °F Vertical Distance Along Furnace Wall, Inches									
	25	26	27	28	29	30	31	32	33	34
40% Primary										
20% Excess	325	313	293	293	282	269	265	250	228	204
40% "	325	314	296	299	289	275	272	255	236	210
60% "	324	312	296	299	292	278	275	259	240	215
80% "	321	310	294	299	291	278	277	259	240	213
100% "	318	309	294	298	290	278	277	259	241	216
60% Primary										
20% Excess	331	319	301	300	289	275	270	255	234	210
40% "	330	319	300	303	292	279	275	259	239	214
60% "	320	309	294	295	287	275	272	257	237	213
80% "	320	309	293	296	288	275	274	258	239	213
100% "	316	307	294	295	288	275	274	258	239	215

Table VII. Quantity of Radiation Heat Transfer and Total Heat Transfer

Aeration	H_2O %	CO_2 %	$T_g$avg °R	$T_w$avg °R	P_c l# Atm-ft	P_w l# Atm-ft	P_c l($\frac{T_w}{T_g}$) Atm-ft	P_w l($\frac{T_w}{T_g}$) Atm-ft
40% Primary								
(1) 20% Excess	17.3	7.84	1660	785	0.0196	0.0432	0.0093	0.0204
(2) 40% "	15.0	6.78	1660	785	0.0169	0.0375	0.0080	0.0178
(3) 60% "	13.4	6.00	1660	785	0.0150	0.0335	0.0071	0.0159
(4) 80% "	12.1	5.35	1660	785	0.0134	0.0302	0.0064	0.0143
(5) 100% "	11.1	4.77	1660	785	0.0119	0.0278	0.0056	0.0132
60% Primary								
(6) 20% Excess	16.9	8.12	1660	785	0.0203	0.0422	0.0096	0.0200
(7) 40% "	14.8	6.80	1660	785	0.0170	0.0370	0.0080	0.0175
(8) 60% "	13.3	5.83	1660	785	0.0146	0.0332	0.0069	0.0157
(9) 80% "	12.0	5.45	1660	785	0.0136	0.0300	0.0064	0.0142
(10) 100% "	11.0	4.95	1660	785	0.0124	0.0275	0.0059	0.0130

#l = l x D (5, Chapter 19, p. 691)

Table VII. Quantity of Radiation Heat Transfer and Total Heat Transfer (cont'd)

	ϵ_g	α_g	Radiation Heat Transfer BTU/hr	Heat Transfer BTU/hr	Percentage of Heat Transfer by Radiation
(1)	0.0458	0.0738	1590	4020	39.5
(2)	0.0404	0.0660	1440	3900	36.9
(3)	0.0366	0.0618	1300	3775	34.4
(4)	0.0338	0.0564	1200	3640	33.0
(5)	0.0322	0.0534	1140	3525	32.3
(6)	0.0464	0.0740	1550	4050	38.3
(7)	0.0404	0.0660	1440	3915	36.4
(8)	0.0364	0.0618	1290	3775	34.2
(9)	0.0339	0.0564	1200	3675	32.7
(10)	0.0325	0.0534	1160	3570	32.5

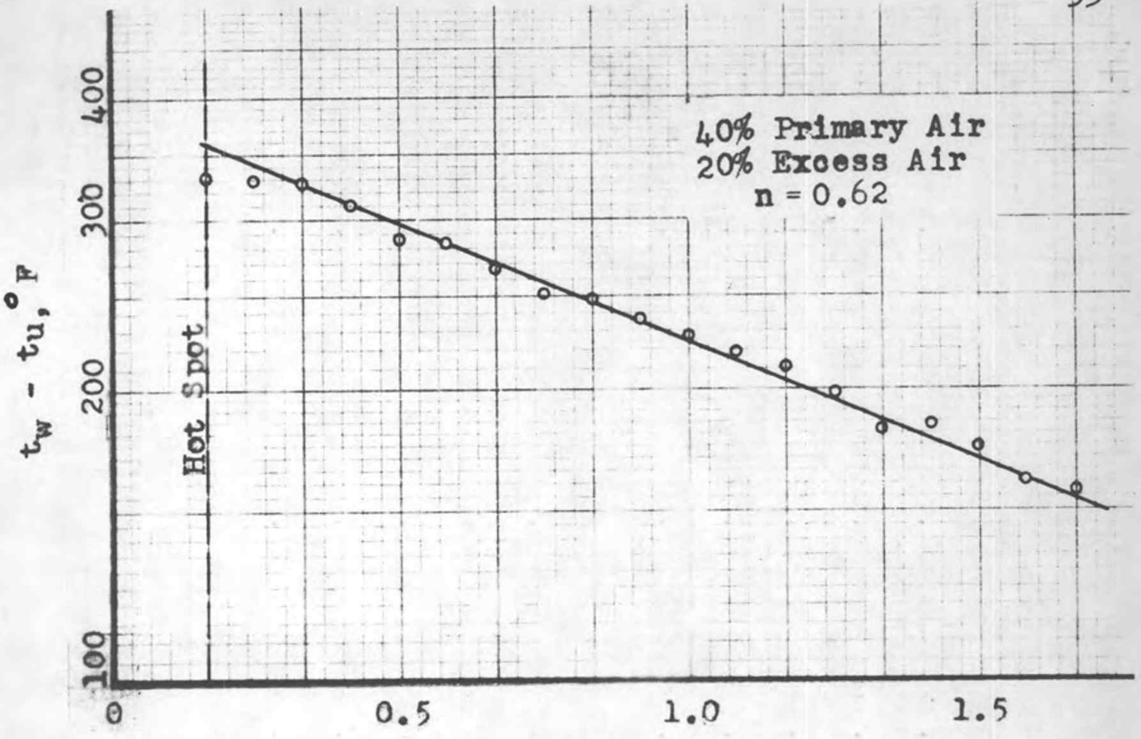
Table VIII. Air Side Heat Transfer Coefficient

Aeration	Average Cooling Air Temp. ^{o}F	Average Furnace Wall Temp. ^{o}F	I - Flue Loss BTU/hr	h_a BTU/hr ft^2 ^{o}F
40% Primary				
20% Excess	105	325	4020	6.15
40% "	105	325	3900	5.97
60% "	105	325	3775	5.78
80% "	105	325	3640	5.56
100% "	105	325	3525	5.40
60% Primary				
20% Excess	105	325	4050	6.20
40% "	105	325	3965	6.00
60% "	105	325	3775	5.78
80% "	105	325	3675	5.63
100% "	105	325	3570	5.46

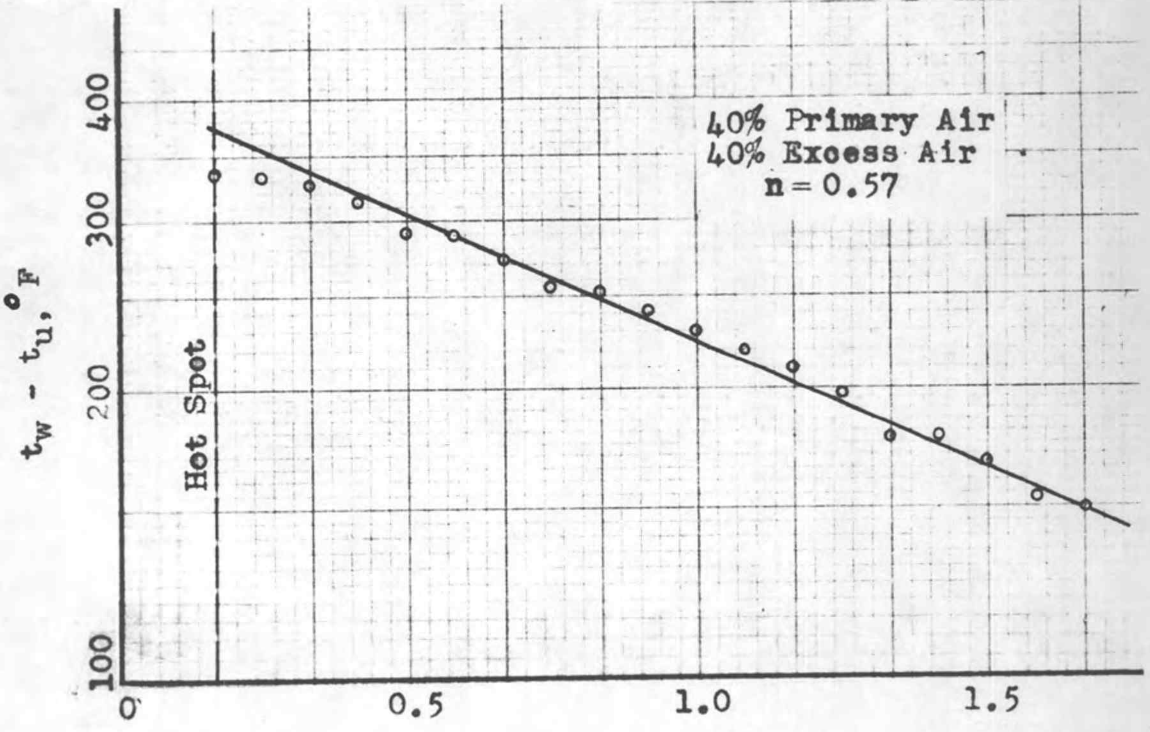
Table IX. Gas Side Heat Transfer Coefficient

Aeration	$w_a c_a$ BTU/hr °F	$w_g c_g$ BTU/hr °F	n# 1/ft	h_a BTU/hr ft^2 °F	h_g BTU/hr ft^2 °F
40% Primary					
20% Excess	62.4	1.46	0.62	6.15	1.38
40% "	62.4	1.65	0.57	5.97	1.43
60% "	62.4	1.83	0.52	5.78	1.48
80% "	62.4	2.02	0.47	5.56	1.48
100% "	62.4	2.23	0.43	5.40	1.51
60% Primary					
20% Excess	62.4	1.43	0.62	6.20	1.35
40% "	62.4	1.65	0.57	6.00	1.45
60% "	62.4	1.87	0.51	5.78	1.48
80% "	62.4	1.99	0.47	5.63	1.49
100% "	62.4	2.16	0.44	5.46	1.49

#Evaluated in Figure 7 - 16

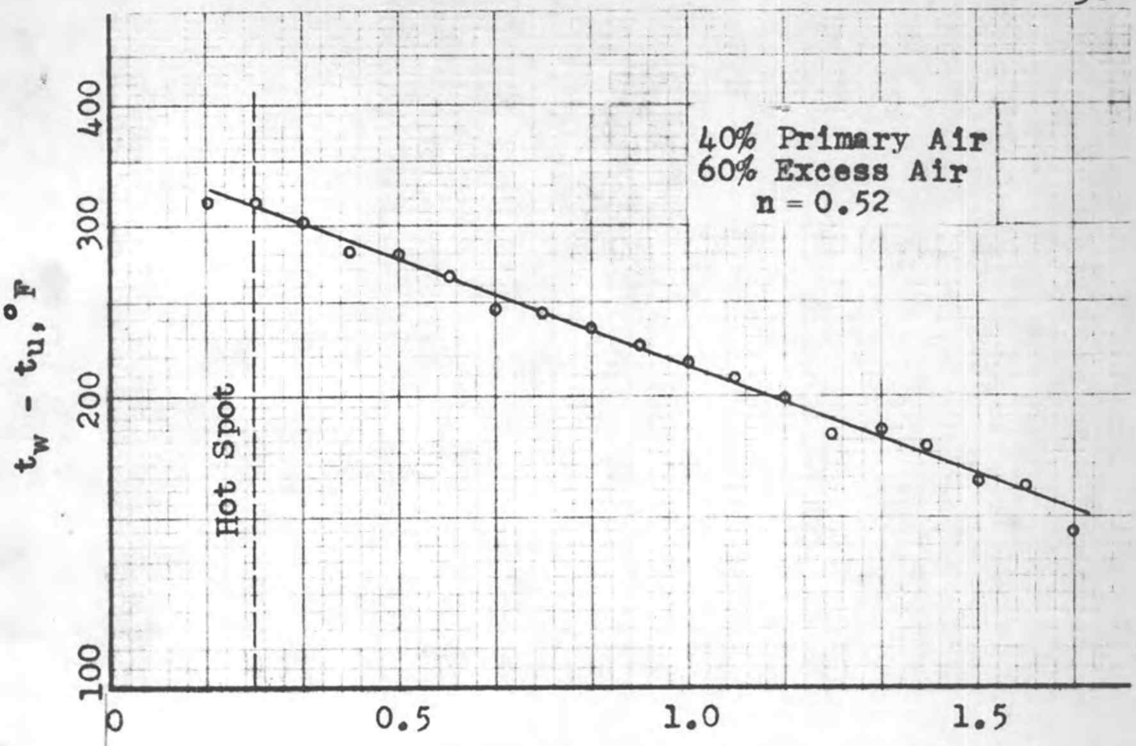


(Fig. 7.) Vertical Distance Above a Reference Level, Ft.

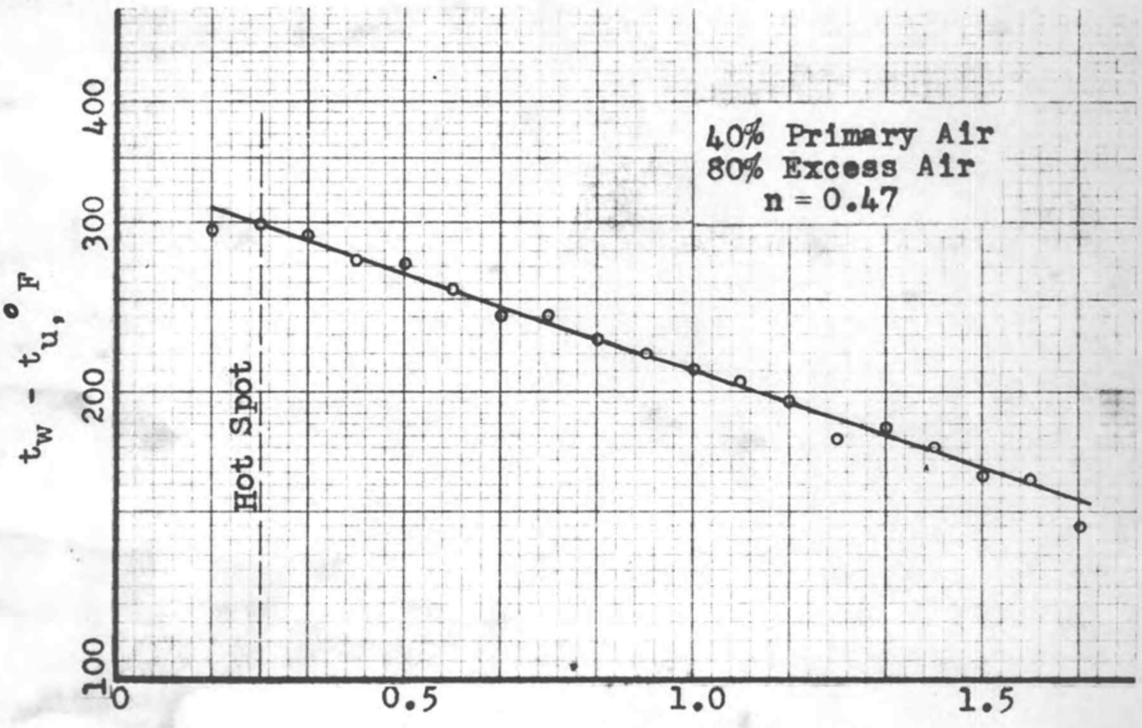


(Fig. 8.) Vertical Distance Above a Reference Level, Ft.

Diagram for Evaluating the Slope of Furnace Wall Temperature Curve in Semi-logarithmic Scale (I) and (II)

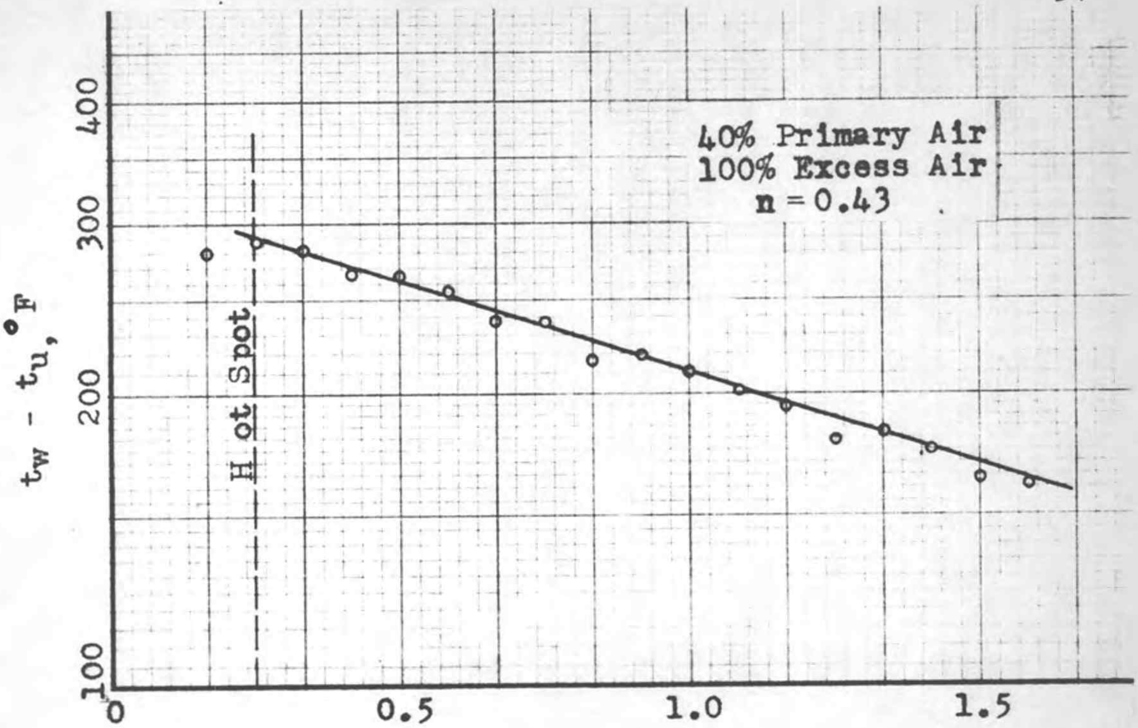


(Fig. 9.) Vertical Distance Above a Reference Level, Ft.

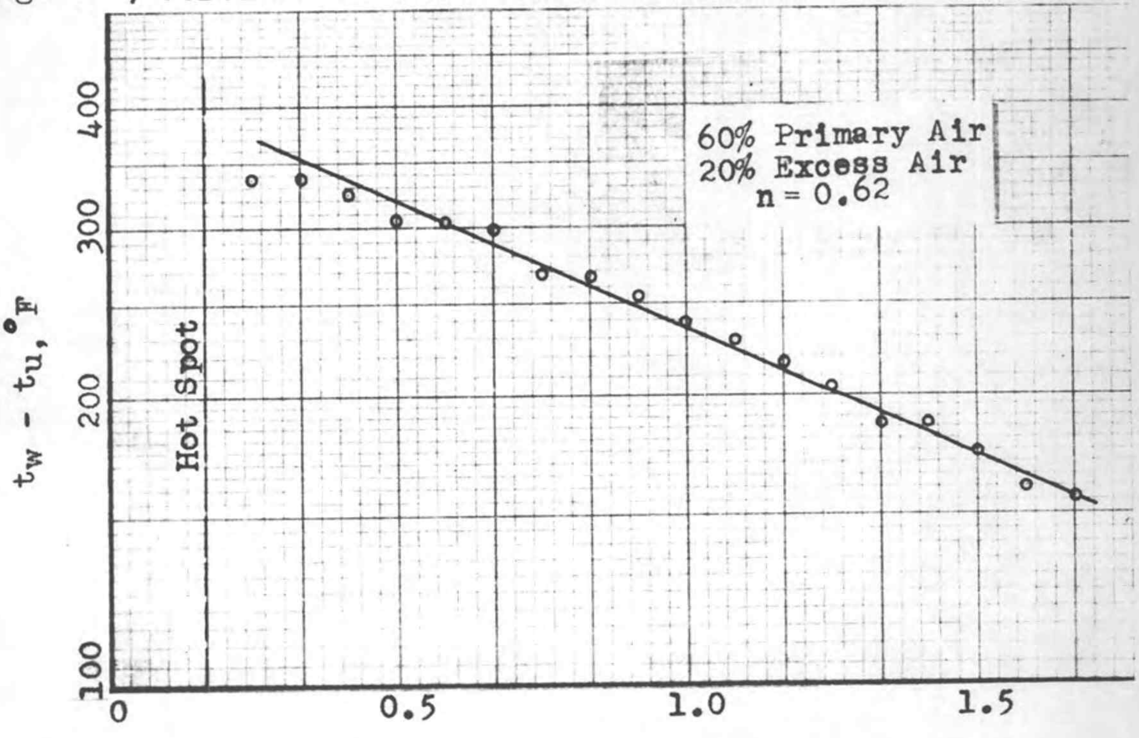


(Fig. 10.) Vertical Distance Above a Reference Level, Ft.

Diagram for Evaluating the Slope of Furnace Wall Temperature Curve in Semi-logarithmic Scale (III) and (IV)

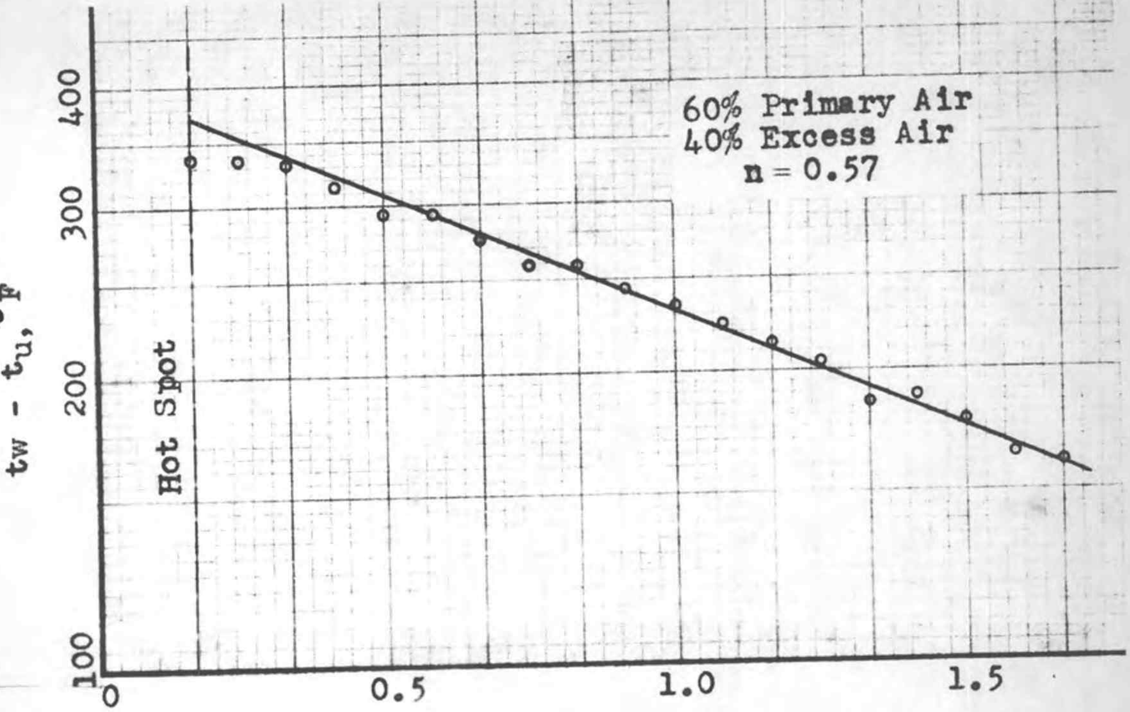


(Fig. 11.) Vertical Distance Above a Reference Level, Ft.

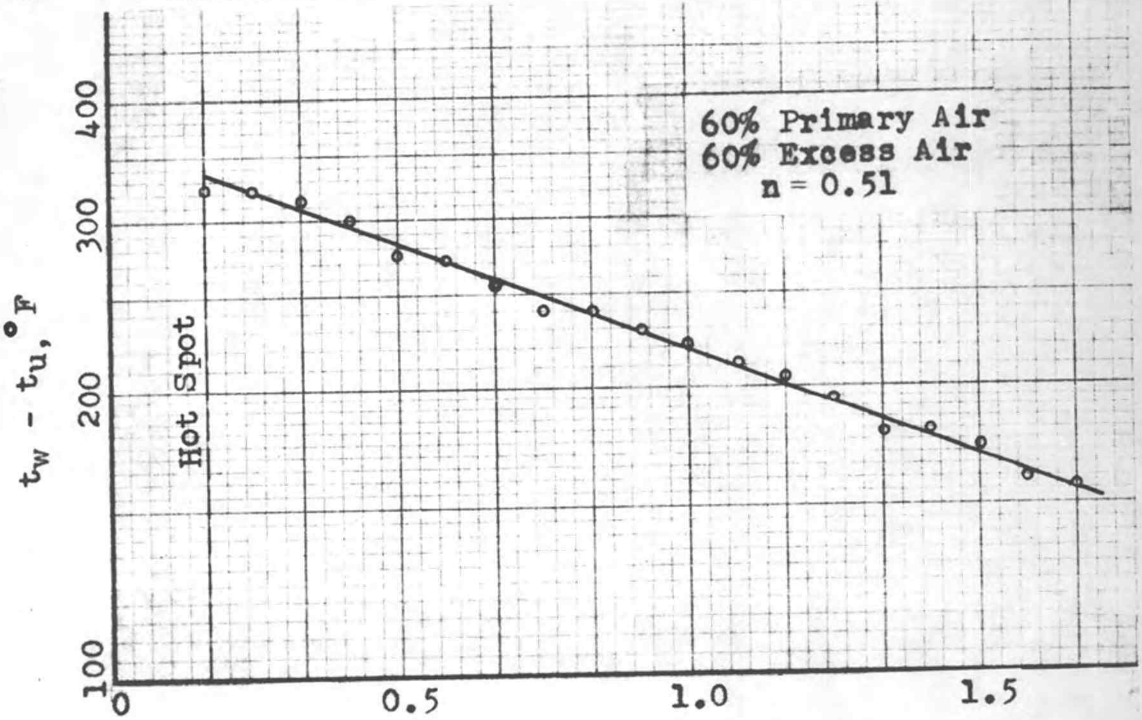


(Fig. 12.) Vertical Distance Above a Reference Level, Ft.

Diagram for Evaluating the Slope of Furnace Wall Temperature Curve in Semi-logarithmic Scale (V) and (VI)

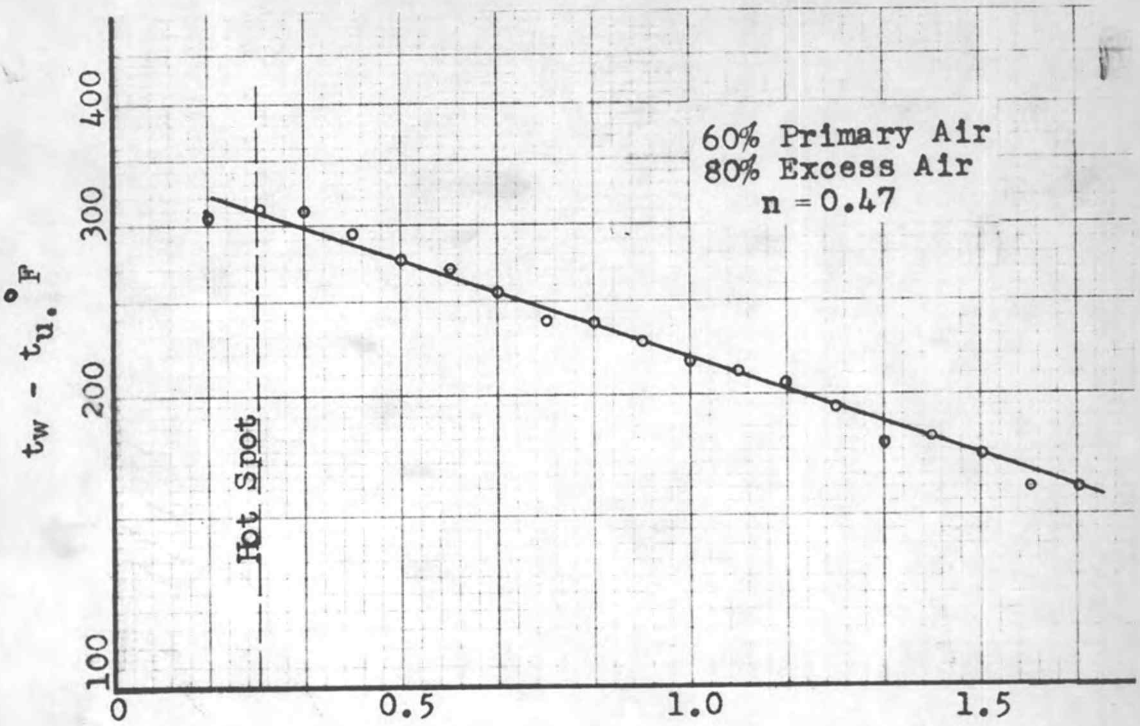


(Fig. 13.) Vertical Distance Above a Reference Level, Ft.

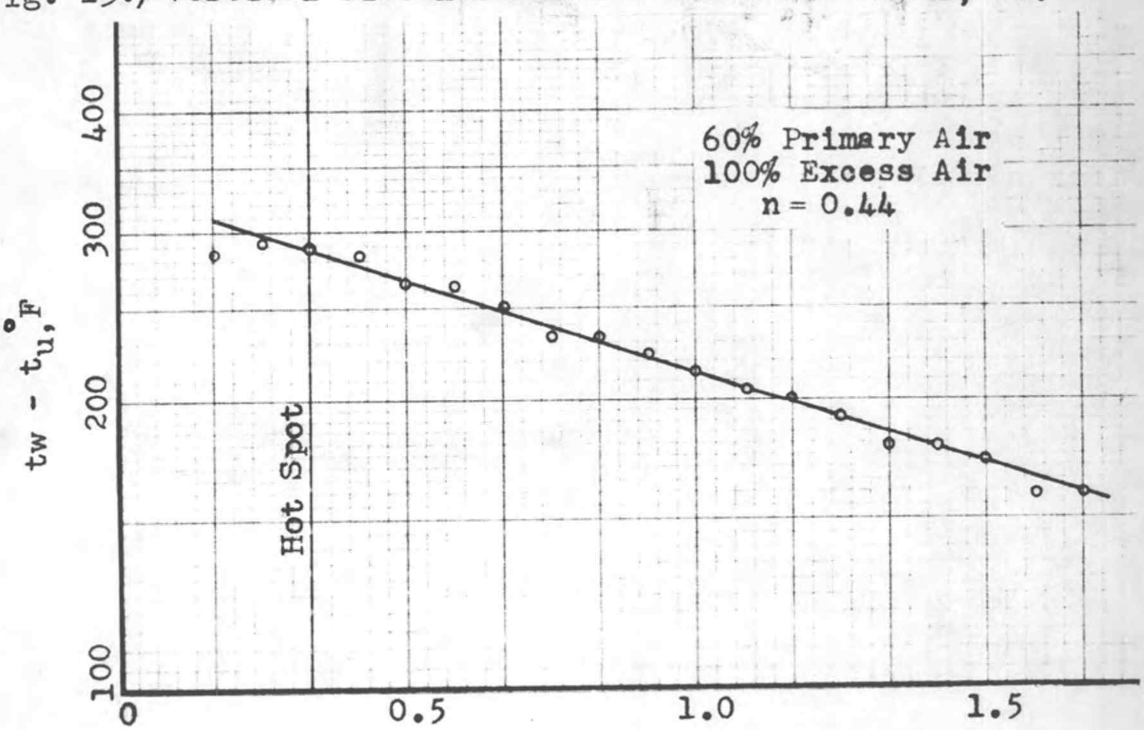


(Fig. 14.) Vertical Distance Above a Reference Level, Ft.

Diagram for Evaluating the Slope of Furnace Wall Temperature Curve in Semi-logarithmic Scale (VII) and (VIII)



(Fig. 15.) Vertical Distance Above a Reference Level, Ft.



(Fig. 16.) Vertical Distance Above a Reference Level, Ft.

Diagram for Evaluating the Slope of Furnace Wall Temperature Curve in Semi-logarithmic Scale (IX) and (X)

Table X. Overall Heat Transfer Coefficient

Aeration	h_a BTU/hr ft^2 °F	h_g BTU/hr ft^2 °F	U BTU/hr ft^2 °F
40% Primary			
20% Excess	6.15	1.38	1.13
40% "	5.97	1.43	1.15
60% "	5.78	1.48	1.18
80% "	5.56	1.48	1.17
100% "	5.40	1.51	1.18
60% Primary			
20% Excess	6.20	1.35	1.11
40% "	6.00	1.45	1.17
60% "	5.78	1.48	1.18
80% "	5.63	1.49	1.18
100% "	5.46	1.49	1.17

CHAPTER VI

Analysis of Data

Analysis of the experimental and calculated data obtained in the present study was divided into three parts; the effect of excess air of furnace wall temperature, the effect of excess air on flue gas temperature, and the effect of excess air on heat transfer and heat transfer coefficients.

Effect of Excess Air on Furnace Wall Temperature A variation of furnace wall temperature for different amounts of excess air is indicated in Figures 17 and 18. The temperature increases slowly from the bottom of the furnace wall to about two inches above the burner port level. This is known as the radiation region as described in the introduction. No obvious change in furnace wall temperature for various amounts of excess air was observed in this region. A small decrement for large excess air was due to some heat being carried away from the furnace wall by the greater amount of secondary air coming from below.

Passing this level, temperature increased rapidly in a length of six inches for 20 per cent excess air to eight inches for 100 per cent excess air until the hot spot or maximum wall temperature is reached. Both convection and

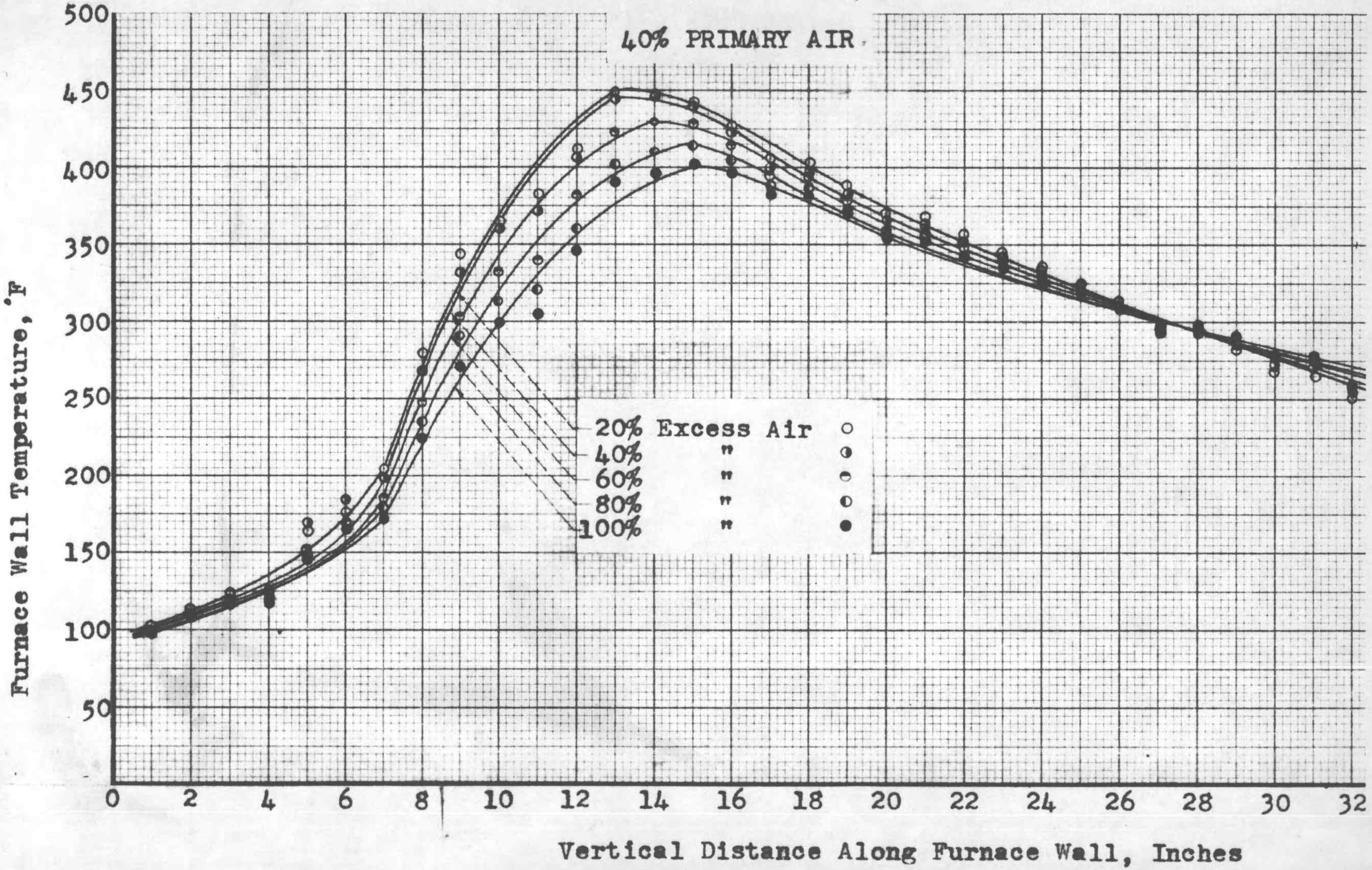


Fig. 17. Furnace Wall Temperature Curve and Its Variation (I)

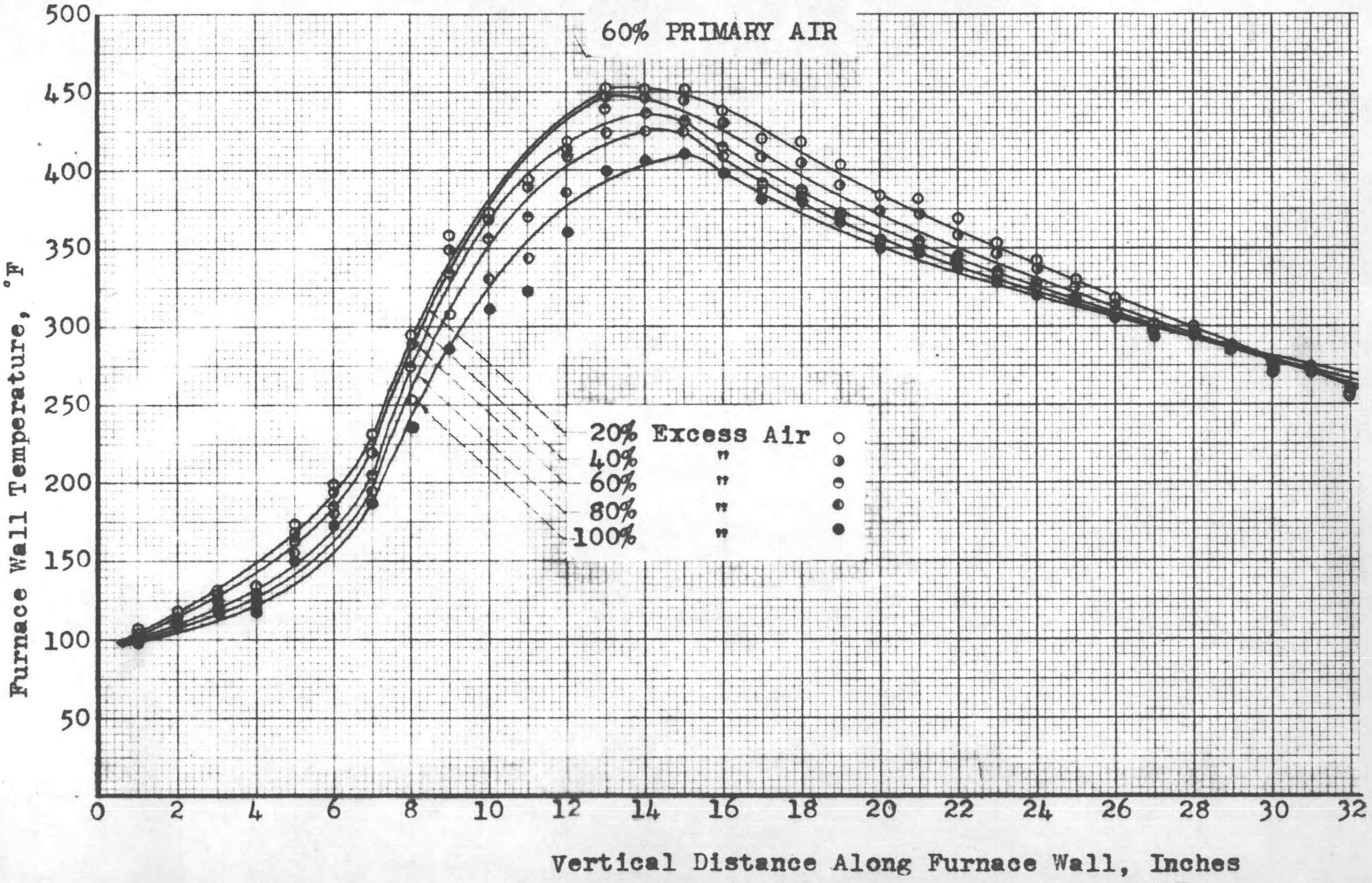


Fig. 18. Furnace Wall Temperature Curve and Its Variation (II)

radiation take place in this so called recirculation region. The slope of wall temperature rise is much greater than that in the radiation region. For a 20 per cent excess air, the slope increases from about 17 °F/in to about 50 °F/in. Increasing the excess air decreases the temperature slope. At 20 per cent excess air the slope is 50 °F/in while at 100 per cent excess air the slope decreases to only 30 °F/in.

Less slope of wall temperature rise in the flame region for higher amount of excess air means a smaller amount of heat transfer in this region. Theoretically, it will cause a lower maximum wall temperature. This is in agreement when plotting the hot spot temperature versus carbon dioxide content in Figure 19. Increasing excess air sometimes causes no effect on hot spot location but other times it moves the location one or two inches upward. Nevertheless, for a certain primary air, increasing the excess air tends to move the hot spot location upward. No change in hot spot temperature and location by changing the primary air from 40 per cent to 60 per cent was observed.

Cooling region furnace wall temperature varies slightly for different amounts of excess air. Large amounts of excess air cause a flatter temperature curve than the smaller amounts of excess air.

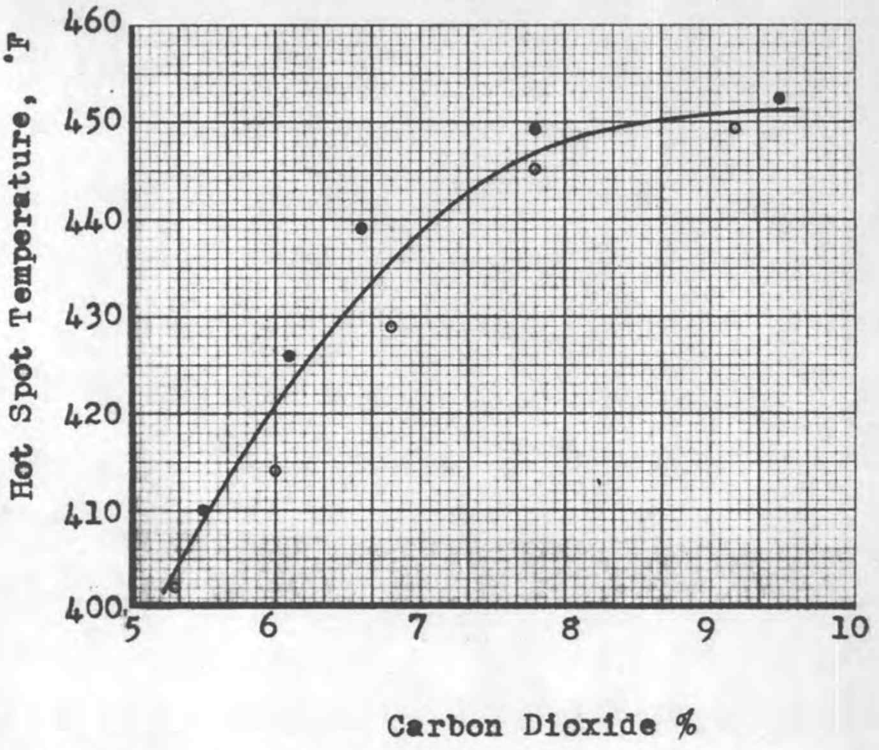


Fig. 19. Variation of Hot Spot Temperature

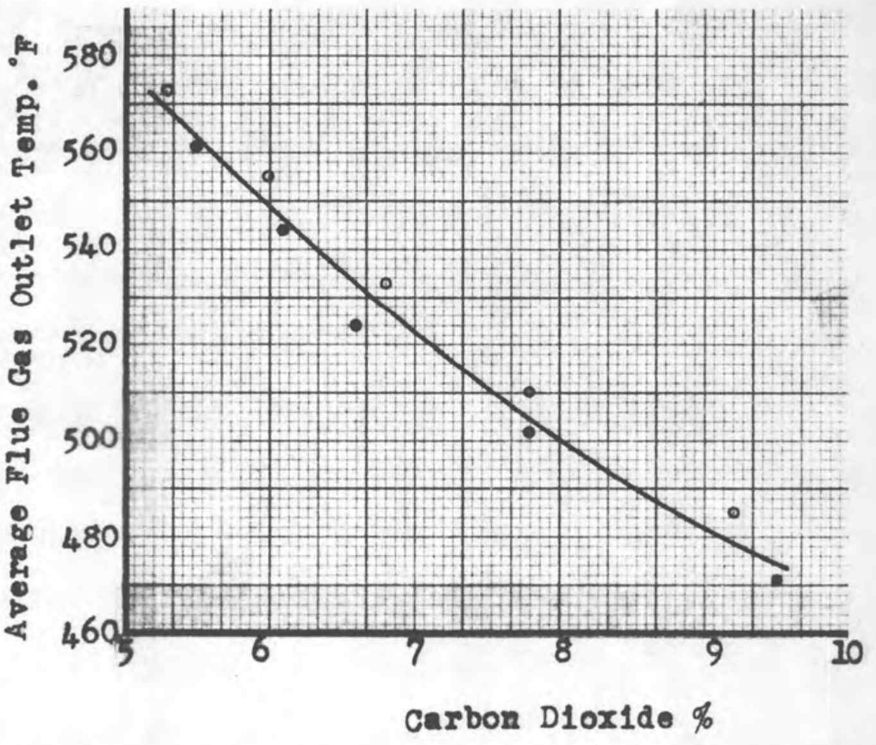


Fig. 20. Variation of Average Flue Gas Outlet Temperature

Effect of Excess Air on Flue Gas Outlet Temperature The flue gas outlet temperature was determined by plotting temperature data provided by the horizontal flue outlet traverse. Temperature distribution for different amounts of excess air is given in Figures 21 and 22. Since the turbulent flow occurs in the flue gas outlet pipe, the weighted average of the flue gas outlet temperature for calculating the heat loss was taken from the distribution curve reading at two-thirds pipe radius from the center without great error. Approximately a 25°F average temperature increase for an increase of 20 per cent excess air could be estimated from both figures. Large amounts of excess air cause smaller percentages of CO_2 and H_2O content in combustion gas. It reduces the heat transfer to the furnace wall and the flue gas outlet temperature increases accordingly. A correlation between the average flue gas outlet temperature and carbon dioxide content is given in Figure 20.

Effect of Excess Air on Heat Transfer and Heat Transfer Coefficients The total amount of heat transfer was simply the heat input minus the heat losses in the flue gas. By plotting the total amount of heat transfer versus carbon dioxide content, a correlation can be obtained as shown in Figure 23. The dotted line in the figure shows

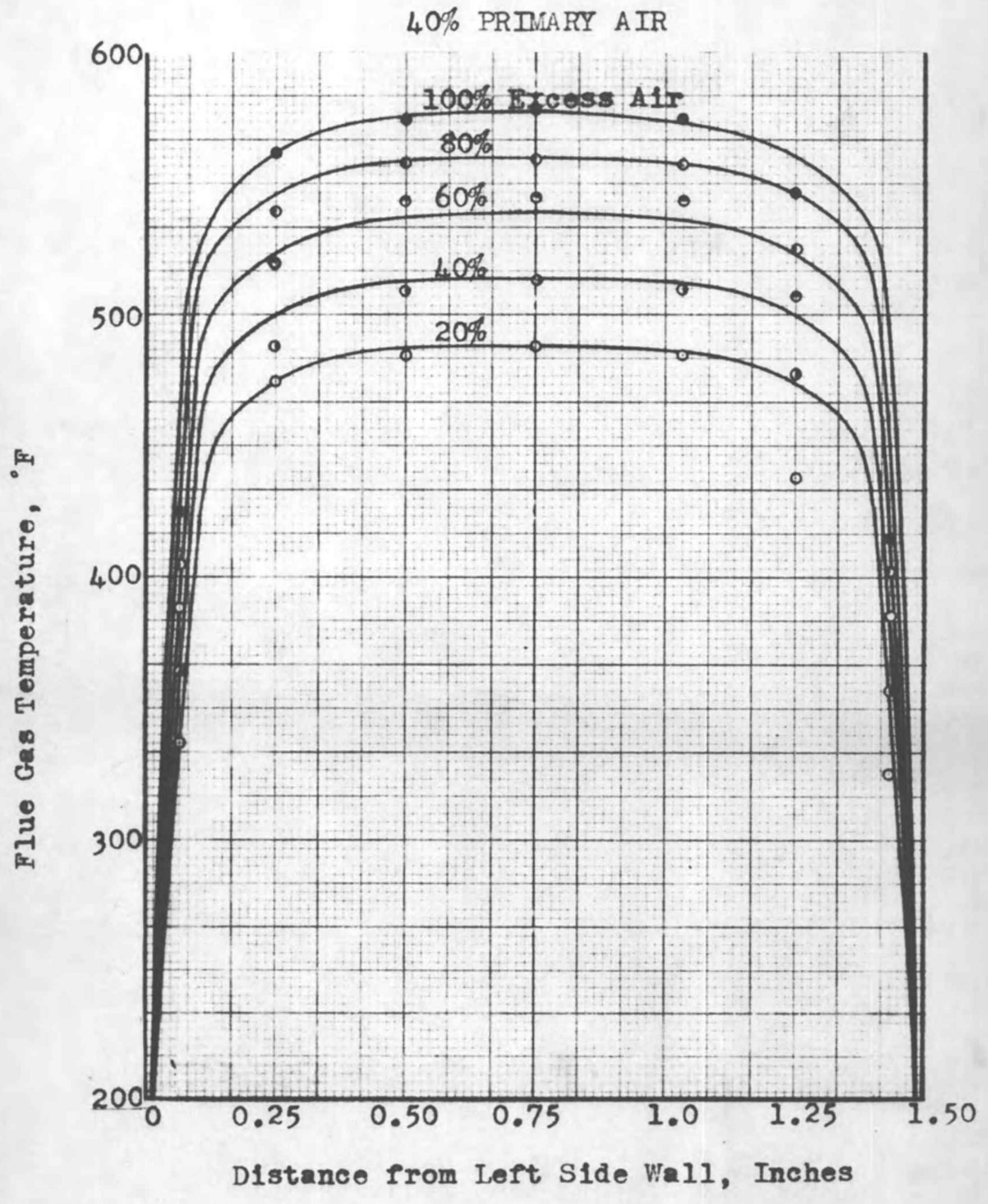


Fig. 21. Flue Gas Outlet Temperature Curve and Its Variation (I)

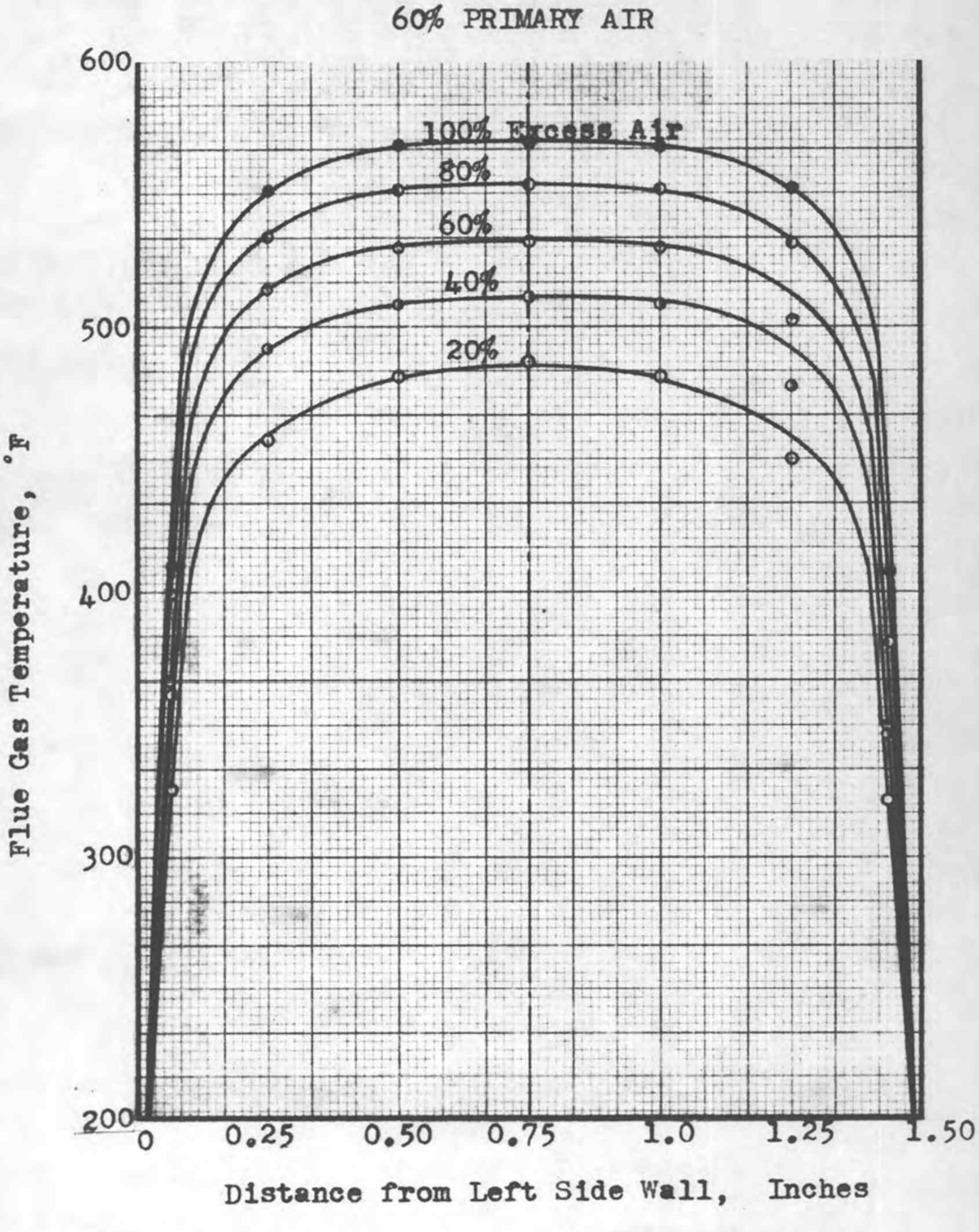


Fig. 22. Flue Gas Outlet Temperature Curve and Its Variation (II)

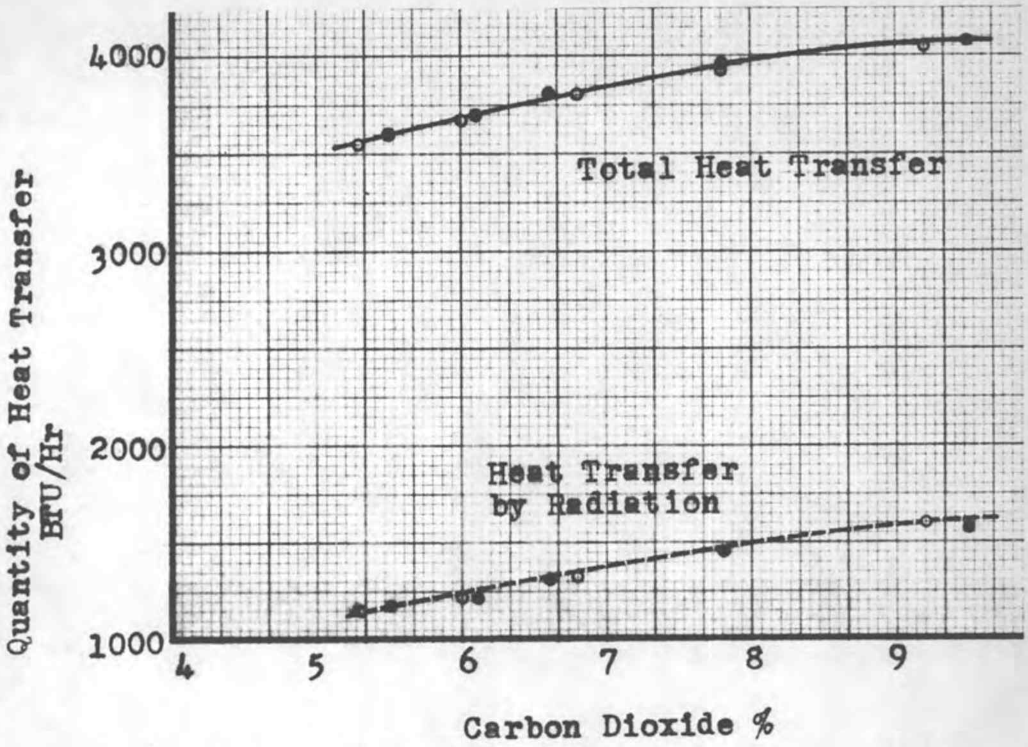


Fig. 23. Variation of Radiation and Total Heat Transfer

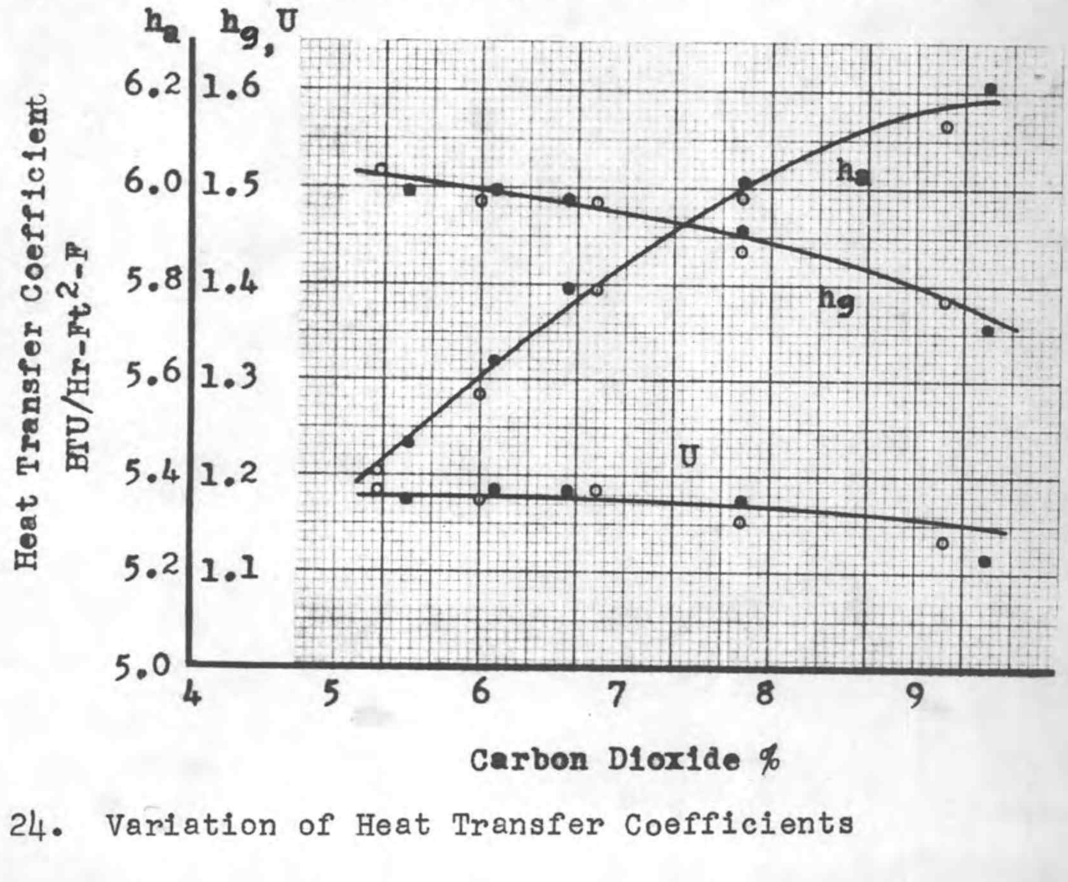


Fig. 24. Variation of Heat Transfer Coefficients

the amount of heat transfer by radiation alone. The amount of heat transfer by convection is the difference of ordinate between two curves. It is seen that both radiation heat transfer and total heat transfer decrease with smaller percentages of carbon dioxide content or an increase in the amount of excess air. The values of percentage of heat transfer by radiation for different amounts of excess air are given in Table VII. A six per cent difference was calculated between 20 per cent and 100 per cent excess air. This was smaller than was expected. The reason might be that the calculated data was based on the same temperature difference between combustion gas and furnace wall for various amounts of excess air. It was actually not true since larger amount of excess air should give a smaller temperature difference.

The variation of the heat transfer coefficients was one of the interesting factors of the present work. Figure 24 shows a correlation between these coefficients and carbon dioxide content. Air side heat transfer coefficient decreases with increase of carbon dioxide content while gas side heat transfer coefficient increases slightly with overall heat transfer coefficient remaining unchanged. The increase of gas side heat transfer coefficient was probably due to higher Reynold's Number of the system causing more turbulence in flow and thus increasing heat transfer by convection. The total quantity of heat transfer depends upon the overall heat transfer coefficient, the area of furnace wall ant the temperature difference between the combustion gas and cooling air. Since the area was unchanged for a given furnace and the temperature difference between the combustion gas and cooling air was reduced for large amount of excess air, the same value of overall heat transfer coefficient would also give a smaller amount of total heat transfer.

CHAPTER VII

Recommendations

In concluding the present study, several investigations are suggested which arise as a consequence of the present study.

1. The results concluded in the present study provide the general information on the problem of heat transfer for various CO_2 content in the flue gas. No data on the change of flame and combustion gas was obtained. A test program should be undertaken on this in further investigations to approach a more reasonable result. Also, a further measurement on the cooling air temperature by installing thermocouple on the air jacket wall is recommended.
2. In order to obtain a result of wide range, an extensive experimental condition is necessary. The heat input can be changed in a range of 5000 BTU/hr to 20000 BTU/hr or more. A large range of excess air is also desired.
3. Shape and dimension of furnace, form and size of burner as well as the type of fuel used are all important variables requiring further investigation. A great deal of information is needed

to bring the problem into more general, applicable and understandable form.

4. At the present time it is necessary to make a number of simplifying assumptions for mathematical solutions of heat transfer and flow phenomena. In order to make precise calculations further studies of chemical radiation from flames and flame flow patterns are necessary in addition to more complete analysis of furnace wall temperature curves.

NOMENCLATURE

A	Effective heat transfer area of furnace wall, ft^2
α_c	Absorptivity of carbon dioxide at T_w and $p_c l(\frac{Tw}{Tg})$, dimensionless
α_g	Absorptivity of combustion gas at T_w, dimensionless
α_w	Absorptivity of water vapro at T_w and $p_c l$ $(\frac{Tw}{Tg})$, dimensionless
c_a	Specific heat of cooling air, BTU/lb ^oF
c_g	Specific heat of combustion gas, BTU/lb ^oF
c_w	Correction factor for the absorptivity and emissivity of water vapor, dimensionless
D	Diameter of furnace, ft
ϵ_c	Emissivity of carbon dioxide at T_g and $p_c l$, dimensionless
ϵ_g	Emissivity of combustion gas at T_g, dimensionless
ϵ_w	Emissivity of water vapor at T_g and $p_w l$, dimensionless
$\epsilon_{w'}$	Effective emissivity of furnace wall, dimensionless ($\epsilon_{w'} = 1$ was taken in the present work)
h_a	Air side heat transfer coefficient, BTU/hr ft^2 ^oF
h_g	Gas side heat transfer coefficient, BTU/hr ft^2 ^oF
I	Total heat input, BTU/hr
l	Radiation mean beam length, ft
n	Slope of furnace wall temperature curve in semi-logarithmic scale, 1/ft

p_c	Partial pressure of carbon dioxide, atmosphere
p_w	Partial pressure of water vapor, atmosphere
Q	Rate of heat transfer from combustion gas to cooling air, BTU/hr
Q_r	Rate of heat transfer by radiation, BTU/hr
t_a	Mean cooling air temperature, ^oF
t_g	Mean combustion gas temperature, ^oR
t_u	Ultimate system temperature, ^oF
t_w	Mean furnace wall temperature, ^oF
t_w	Mean furnace wall temperature, ^oR
U	Overall heat transfer coefficient from combustion gas to cooling air, BTU/hr ft^2 ^oF
w_a	Weight of cooling air, lb/hr
w_g	Weight of combustion gas, lb/hr

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A P P E N D I X

APPENDIX

Equations Used in the Calculations

Percentage of carbon dioxide content on wet basis was calculated by

$$CO_2 \text{ (wet basis)} = CO_2 \text{ (dry basis)} \times \frac{100}{100 + H_2O} \qquad (5)$$

Emissivity of combustion gas was calculated by

$$\epsilon_g = \epsilon_c + C_w \epsilon_w \qquad (6)$$

Absorptivity of combustion gas was calculated by

$$\alpha_g = \alpha_c \left(\frac{Tg}{Tw}\right)^{0.65} + C_w \alpha_w \left(\frac{Tg}{Tw}\right)^{0.45} \qquad (7)$$

Product of weight and specific heat of combustion gas was calculated by (1, p. 150)

$$w_g c_g = \frac{I}{10000} \left(\frac{19.2}{CO_2} + 0.835\right) \qquad (8)$$

Ultimate system temperature was calculated by

$$t_u = \text{Room temperature} + \frac{I}{w_a c_a + w_g c_g} \qquad (9)$$